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AREA THERM

THERMAL SUPPRESSION OF EXHAUST PIPES

FINAL REPORT
ATC-55

By

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Under Contract to

U.S. Army Mobility Equipment
Research and Development Command
Fort Belvoir, Virginia

Contract No. DAAK70-82-C-0040

Approved for public release; distribution unlimited

September 1982

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER ATC 55	2. GOVT ACCESSION NO. ADA119455	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) Thermal Suppression of Exhaust Pipes		5. TYPE OF REPORT & PERIOD COVERED Final Report Feb. 1982 – Aug. 1982
		6. PERFORMING ORG. REPORT NUMBER ATC Report No. 55
7. AUTHOR(s) Edward F. Allard		8. CONTRACT OR GRANT NUMBER(s) DAAK70-82-C-0040
9. PERFORMING ORGANIZATION NAME AND ADDRESS U.S. Army Mobility Equipment Research & Development Command Attn: DRDME-M Ft. Belvoir, VA 22060		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS 1L162733AH20-EE
11. CONTROLLING OFFICE NAME AND ADDRESS		12. REPORT DATE September 1982
		13. NUMBER OF PAGES 31
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release;   distribution unlimited

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

entrainment
thermal suppression
exhaust

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

Under the above contract, we were successful in thermally
suppressing the hot exhaust gas on a 3 kW engine-
generator set. This realistic experiment cooled the
engine's exhaust gas from around 400°C down to around
55°C.

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SUMMARY

Under a contract with MERADCOM, Fort Belvoir, Virginia,
Area Therm Corporation has documented the use of an air
ejector for thermal suppression applications. The results
show that an engine's exhaust pipe with a temperature near
400°C can be effectively reduced to a temperature near 50°C.
This is a temperature reduction of 350°C. When one considers
that such performance was achieved with a single pipe and a
nozzle on the engine's exhaust pipe, the results are astoun-
ding. Furthermore, these results were obtained from the very
first design. Clearly, newer designs have the possibility of
even better results. ATC utilized a simplified theory, devel-
oped for this program, as a guide to design. Utilizing known
data from the technical literature on certain sections of the
design, an air ejector was designed, built and tested. Entrain-
ment ratios were measured for this ejector using 0.5" and 0.75"
nozzles. The flow through the nozzles was measured by an air
manifold system. Ambient air, heated air and exhaust gases
were forced through the nozzles, and entrainment ratios were
measured for these driving fluids. Entrainment ratios from 6
to 10 were achieved. These ratios were responsible for the
temperature reductions. The most significant results are for
the case of directly feeding the exhaust of a 3kW engine-
generator set into the air ejector. This realistic experiment
cooled the engine's exhaust gas from around 400°C down to
around 55°C. The cooled gas heated the ejector pipe to 50°C
which consequently resulted in the outstanding thermal suppres-
sion.

TABLE OF CONTENTS

1.0   INTRODUCTION

This report describes the work performed by Area Therm
Corporation under USAMERADCOM Contract No. DAAK70-82-C-0040.
The work was performed at ATC between 5 February 1982 and
15 August 1982.

1.1   Project Objectives

The two primary objectives of this program were:

(1). Develop a theory to help as a guideline for
      air ejector designs

(2). Build and test an air ejector and test the
      ejector on a 3kW engine-generator set

1.2   Methodology

A theoretical expression was developed, based on funda-
mental air flow phenomena. The theory revealed that air entrain-
ment ratio was quadratically dependent on the ratio of the area
of the nozzle and the area of the mixing tube in the ejector.
The greater this area ratio, the greater the entrainment ratio.
In addition, the theory revealed that certain flow efficiencies
of nozzle and "bell mouth input" directly affected entrainment
ratio.

An air flow manifold was built to measure the volume of
air flowing through the nozzles under test. An orifice plate
was used to measure the driving fluid, Q_1, through the nozzle.
A valve was used to control the flow from a high pressure
blower. See Figure 6 . The entrained air, Q_2, was measured
by use of a pitot tube which measured both total pressure and
static pressure.

The nozzle and air ejector were hogged out of solid
aluminum for these experiments. The nozzle design was a scaled
up version of a nozzle published by the National Bureau of
Standards. The "bell entrance mouth" was a scaled up design of
the same NBS paper. The mixer and diffuser designs were based
on theoretical considerations and known data on diffusers.

1.3 Outline of Report

This report has been organized into the following sections:

Section 2 describes the theoretical considerations con-
cerned with air ejectors and defines terms.

Section 3 describes the design of the nozzles and the
design of the air ejector.

Section 4 describes the test apparatus.

Section 5 presents the data.

Section 6 states the results.

Section 7 discusses the results and conclusions to be
drawn.

Section 8 presents a short technical discussion.

Section 9 recommends future work.

2.0 Theoretical Considerations

It has been difficult historically to accurately predict
the performance of air flow systems. For this reason, much of
the work has been experimental in nature. For the most part,
theory is used to guide the designer. In every case, the
designer tests his design and tries to correlate his test
results with theory. The experiments in this report are no

different. A simple theory was developed to guide the design
of an air ejector. The air ejector was built with the guide
in mind and then tested. In order to understand the technical
issues at hand, several technical definitions are given for
the reader's convenience. An air ejector is an air pump. A
driving fluid is forced through a nozzle where it is directed
into the entrance of an air ejector. The energy in the driving
fluid is used to draw ambient air into the entrance where it
enters the mixing section of the ejector. In the mixing section,
the driving fluid is mixed with the entrained air. If the driving
fluid is a hot gas, the mixing cools the hot gas. From the mixing
tube, the air-fluid mixture flows into an expander or diffuser
section. Proper design of the diffuser increases the entrainment.
From the diffuser, the mixture flows into a straight pipe which
would be the exhaust pipe of an engine. For clarity, this pipe
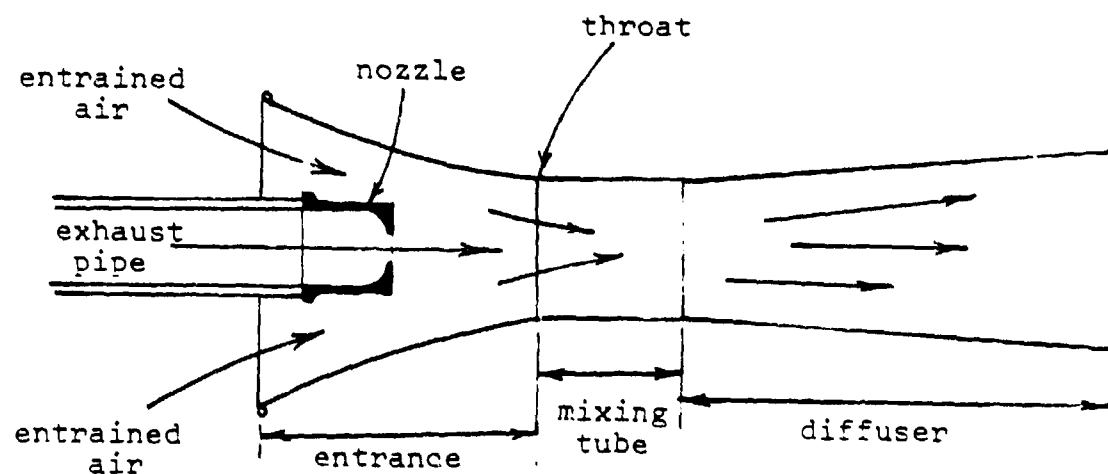is called the ejector pipe. Consider the following diagram.

Figure 1

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nozzle - A section attached to an exhaust pipe designed to provide a known configuration for the primary gas stream.

entrance - That section of the air ejector through which air enters the mixing tube. The bell shaped mouth is designed to reduce air friction between the entrained air and the entrance to the mixing tube.

mixing tube - A straight section of pipe where the primary gas from the engine is mixed with entrained air.

diffuser - A slowly diverging section of pipe which allows the air-gas mixture to expand slowly. This section converts kinetic energy of the air-gas mixture into static pressure.

The measure of "goodness" of an air ejector for present purposes is the amount of air the ejector entrains. The cooling of an exhaust gas depends on the amount of entrained air that is mixed with exhaust gases in the mixing tube. Entrainment ratio is the most important measure, and is defined as

$$\text{Entrainment ratio } N = \frac{\text{volume of entrained air}}{\text{volume of exhaust gas}}$$

The objective of any design effort is to increase the entrainment ratio. Thus, two ejector designs can be compared by simply measuring their entrainment ratios.

There are many parameters that can influence entrainment ratio. Some of the parameters are listed below to illustrate something of the magnitude of the problem:

.exhaust gas rate of flow

.exhaust gas pressure

.specific gravity of the exhaust gas

.exhaust gas temperature

.configuration of exhaust gas after leaving the nozzle

.specific gravity of entrained air

.temperature of entrained air

.atmospheric pressure

.nozzle design

.position of the nozzle's orifice relative to the
   throat of the mixing tube

.design of the entrance section

.diffuser design

.backpressure in the diffuser

.friction between the gases and the wall of the
   ejector

2.1 Momentum

It is useful to describe certain phenomena verbally before
reducing the relationships to mathematical expressions. Thus,
the momentum of a volume of gas is equal to its mass times its
velocity. The momentum of gas, leaving the orifice of a nozzle,
can be measured. This momentum performs two functions:

    (1). It provides the necessary force to drive
        the air-gas mixture through the ejector.

    (2). It provides the necessary force to entrain
        the air into the mixer.

The design problem is to convert as much of the gas momentum
into entrainment force as possible, and still have sufficient
force remaining to drive the mixture through the ejector. The
quantity of entrained air will depend on the velocity of the
exhaust gas, the specific gravity of the gas and the rate of
flow. Of course, there is always some loss in momentum due to
eddy currents and friction. Therefore, the momentum of the air-

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gas mixture will be less than the momentum of the exhaust gas.
It can be seen from experience that more force is required to
drive a mixture through a straight pipe than through a pipe
that gradually expands. Therefore, a diffuser makes it easier
⁴⁾ drive the mixture through an ejector and, in turn, permits
more gas momentum to be used for entrainment.

2.2 Flow Principles

There are several flow principles that should be defined
in order to illustrate the issues at hand. Insofar as the gas
and air-gas flows are always turbulent in this type of configur-
ation, and recognizing that there are no great pressure drops
capable of causing shock waves or hypersonic waves, simplier
equations can be used with sufficient accuracy. The basic
relation underlying fluid flow is called the hydraulic equation:

$$v = \sqrt{2gh}$$

where v = velocity of the fluid (feet/sec)

      g = acceleration due to gravity

      h = pressure head of the flowing fluid

The last equation can be rewritten as

$$h = \frac{v^2}{2g}$$

This head is referred to as the velocity head. It is the pres-
sure head required to produce the momentum of the fluid. Since
the pipe has friction, some of the pressure head will be lost
and it becomes necessary to increase the pressure head in order

to maintain the original momentum. For example, if the friction causes a loss of 1/2 a velocity head, then the total pressure head required would be

$$h = (1 + \tfrac{1}{2})\frac{v^2}{2g}$$

Note that the loss due to friction is measured in terms of the original velocity head. If f denotes the total head loss due to friction then

$$h = (1 + f)\frac{v^2}{2g}$$

This expression can be put in a more convenient form when one recalls that the volume of fluid flowing through an area A is

$$Q = vA$$

$$\text{or}\quad v = Q/A$$

$$h = (1 + f)\frac{Q^2}{2gA^2}$$

$$\text{or}\quad Q = A\sqrt{\frac{2gh}{(1 + f)}} \quad (ft^3/sec)$$

Since many pressure measuring devices are in increments of inches of water, this last expression can also be expressed by rewriting h as

$$h = \frac{H}{12}\,\frac{820}{d}$$

where H = inches of H_2O

   d = the specific density of gas at $60^{\circ}F$

   820 = $\dfrac{\text{density of water at } 60^{\circ}F}{\text{density of air at 30 inches Hg and } 60^{\circ}F}$

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$$Q = A \sqrt{64.4 \times 68.33 \frac{H}{d(f+1)}} = 66.33 A \sqrt{\frac{H}{d(f+1)}} \; \frac{ft^3}{sec}$$

or $$\boxed{Q = 66.33 A \sqrt{\frac{H}{nd}}}$$ where $n = f + 1$ Eq. (1)

In flow through pipes, the friction term depends on the length of pipe, area, roughness, and velocity of the gas. For example, one expression for turbulent flow suggests

$$f = \frac{0.316}{Re^{0.25}}$$

where R_e = the Reynolds Number

In other words, it is possible to measure the friction effect in lost pressure heads for various pipe configurations.

Another way to express pressure head loss due to friction effects is to consider all of the various resistances to flow as a single resistance term. For example, entrances to a duct, exits, side openings, expansions and contractions, all add friction effects. Therefore, an _effective_ head can be described as

$$\boxed{Q = 66.33 A \sqrt{e\frac{H}{d}}}$$

Eq. (2)

where e = the fractional part of the total head that
is effective at producing flow. The rest
of the head is used up in overcoming the
total resistance.

This expression is useful in describing the performance of a nozzle or the outlet of a blower. A percent of the blower's velocity head is converted into static pressure. The static pressure is used to force the fluid into various outlets.

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There is a third way of expressing the volume of fluid flow that is also useful

$$Q = 66.33 \ cA\sqrt{\frac{H}{d}}$$

Eq. ③

where c = the coefficient of discharge

This expression applies when flow is measured by an orifice or nozzle. The value of c is usually given by an orifice manufacturer or it can be measured independently. When a stream leaves an orifice, the area of the stream contracts (vena contracta), and the value of c corrects for this change in area. So, to measure fluid flow through a pipe, an orifice with a known c and area can be used. All that is required then is to measure the pressure difference before and after the orifice. This is the technique ATC will use in the course of the present contract.

2.3 The Air Ejector

Since entrainment ratio is the key to cooling an exhaust pipe, it is necessary to measure the performance of the ejector itself. Consider an ejector for exhaust pipe suppression
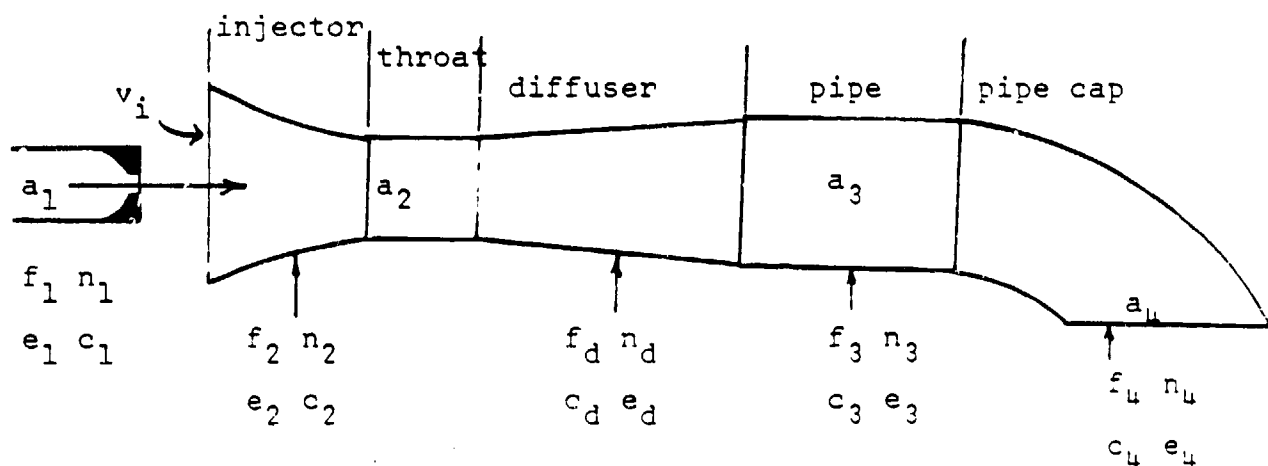

Figure 2

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Each section of the ejector can be considered from the viewpoint
of friction, efficiency, or flow coefficient. Therefore, each
section has an f, n, c and e, as shown in the figure. From
equations ①, ② and ③, the following relations can be
recognized among f, n, e and c:

$$f = n - 1 = \frac{1}{c^2 - 1} = \frac{1}{e} - 1$$

$$n = \frac{1}{e} = \frac{1}{c^2} = f + 1$$

$$e = c^2 = \frac{1}{n} = \frac{1}{1 + f}$$

$$c = \sqrt{e} = \sqrt{\frac{1}{n}} = \sqrt{\frac{1}{1 + f}}$$

Given these relationships, it becomes necessary to add resistances
in series and to derive a single resistance for the whole ejector.
The single number can then be expressed in any of the four resis-
tance terms. The procedure is to add all friction terms, obtain
a total friction and then convert this term into the other expres-
sions for resistance. There is no simple way to add the other
resistance terms. Since the nozzle supplies momentum, there is
one velocity head that can be used as a reference to measure
lost heads in the various sections. However, one can use any
section as a reference point. In the fluid flow literature, some
authors use the initial head as a reference and others use the
final head as reference. Since f, n, e and c are dependent on
velocities and areas of particular sections, the friction term
for each section must be normalized to a reference point. For
example, if there are four sections to an ejector system, and

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the fourth section is taken as the reference section for pressure head and area, then the total friction in terms of velocity head at section 4 is given by

$$f_T \text{ (ref. vel. head at section 4)} = \left(\frac{a_4}{a_1}\right)^2 f_1 + \left(\frac{a_4}{a_2}\right)^2 f_2 + \left(\frac{a_4}{a_3}\right)^2 f_3 + f_4$$

Equation ① shows that friction is proportional to the square of area, thus:

$$n_T = 1 + f_T$$

$$e_T = \frac{1}{1 + f_T}$$

$$c_T = \sqrt{\frac{1}{1 + f_T}}$$

2.4 Ejector Efficiency

An air ejector as described here can be considered from the viewpoint of a Venturi meter. Using this approach, it is possible to derive the flow coefficients for entrained air and thereby define the efficiency of the ejector. For a Venturi meter

$$\sqrt{v_2^2 - v_i^2} = C_2 \sqrt{2g\ \Delta h}$$

where v_i = velocity of stream at inlet of injector (ft/sec)

v_2 = velocity in the throat (ft/sec)

C_2 = flow coefficient of throat based on the throat area

Δh = drop in static head between the inlet to the injector and the throat

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This last expression can be rewritten

$$\Delta h = \frac{1}{C_2^2} \; \frac{v_2^2 - v_i^2}{2g}$$

For our ejector (atmospheric pressure) the particles of air outside the injector are initially at rest so that $v_i = 0$. Then,

$$\Delta h = \frac{1}{C_2^2} \; \frac{v_2^2}{2g} = (1 + f_2) \; \frac{v_2^2}{2g}$$

The coefficients C_2 and f_2 are for the curved surface of the entrance section. A given momentum is required to push the stream through the throat and additional momentum is required to overcome f_2. From another viewpoint, the velocity in the throat provides the suction for entrainment and the necessary force to push the stream through the throat. Of course, the energy or momentum to do these things comes from the flow through the nozzle. The momentum from the nozzle must be used to overcome all the additional friction effects encountered after passing through the throat. The resistance for the diffuser is given by $f_d \frac{v_2^2}{2g}$. The resistance for the straight tube is $f_3 \frac{v_3^2}{2g}$. The resistance for the cap is $f_4 \frac{v_4^2}{2g}$. The total resistance for the ejector is

$$(1 + f_2) \; \frac{v_2^2}{2g} + f_d \; \frac{v_2^2}{2g} + f_3 \; \frac{v_3^2}{2g} + f_4 \; \frac{v_4^2}{2g}$$

If we refer all the velocities (pressure heads) to the velocity in the throat, the total resistance becomes

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$$f_T + 1 + f_2 + f_d + f_3 \left(\frac{a_2}{a_3}\right)^2 + f_4 \left(\frac{a_2}{a_4}\right)^2$$

Eq. (4)

The coefficient f_d takes into consideration the change in area and velocity in the diffuser. Since this last term represents the lost velocity heads, the nozzle must supply these lost velocity heads as well as the velocity head required to push the stream through the ejector. The total number of velocity heads is, then,

$$N_T = 1 + 1 + f_2 + f_d + f_3 \left(\frac{a_2}{a_3}\right)^2 + f_4 \left(\frac{a_2}{a_4}\right)^2$$

Eq. (5)

The fraction of the velocity head in the throat that can be used to push the stream through the ejector is

$$e_T = \left[2 + f_2 + f_d + f_3 \left(\frac{a_2}{a_3}\right)^2 + f_4 \left(\frac{a_2}{a_4}\right)^2 \right]^{-1}$$

Eq. (6)

The flow coefficient for the entire ejector is

$$C_T + \sqrt{\left[2 + f_2 + f_d + f_3 \left(\frac{a_2}{a_3}\right)^2 + f_4 \left(\frac{a_2}{a_4}\right)^2 \right]^{-1}}$$

Eq. (7)

Notice that from equation (6), the effective velocity head will always be less than 50%. From equation (7), the flow coefficient will always be less than 0.707. These equations reveal some interesting things from the viewpoint of cooling an exhaust pipe. If the cap on the pipe introduces a high friction term, the efficiency of the ejector is reduced. A sharply curved cap can drastically reduce the efficiency. Of course, from a cooling viewpoint, it is best

to have no cap at all. The equations show that the area of the
exhaust pipe should be as large and as smooth as possible. In
practice, this term is relatively small. The diffuser should
be designed to have a low friction coefficient. This is an
important term and affects the efficiency more than any other
term, excluding a "bad" cap.

2.5 The Importance of the Ratio of Orifice Area to Throat Area on Entrainment Ratio

Anticipating future results, the ratio of the nozzle opening
to the throat area is the primary limit on the entrainment ratio
up to a point; the greater the area ratio, the greater the
entrainment ratio. The area ratio can be increased by making
either the throat larger or the nozzle opening smaller. Increasing
the throat makes a large ejector pipe, while decreasing the noz-
zle opening could cause an increase in engine back pressure. Con-
sider the following diagram for describing the mixing of a volume
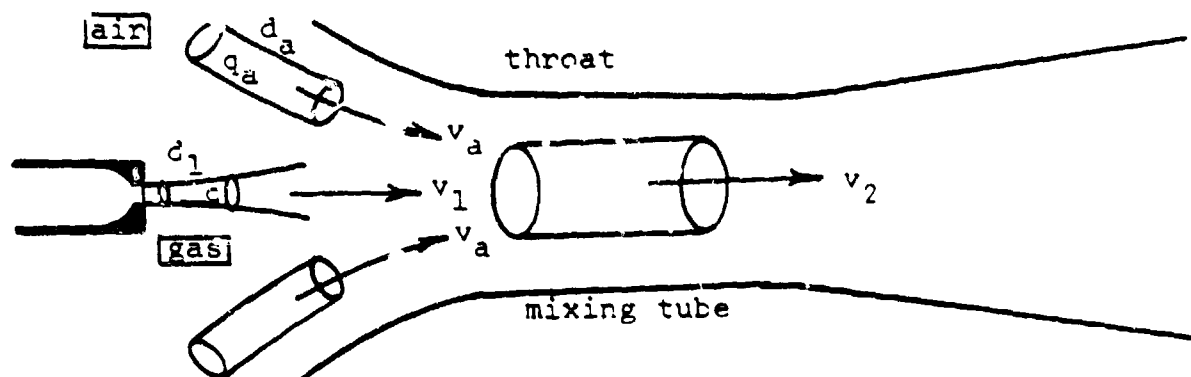of gas from the exhaust pipe with a volume of entrained air.

Figure 3

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The momentum of the gas stream is used to entrain air and then
drive the air-gas mixture through the ejector. Without friction,
conservation of momentum demands that the momentum of the gas
be equal to the momentum of the mixture.

$$q \, d_1 \, v_1 = q(N + 1) \, \frac{(Nd_a + d_1)}{N + 1} \, v_2$$

where q =   volume of gas (ft^3/sec)

  d_1 =   density of gas (lbs/ft^3)

  v_1 =   velocity of gas (ft/sec)

  N =   entrainment ratio

  d_a =   density of air (lbs/ft^3)

  v_2 =   velocity of mixture (ft/sec)

The term $\frac{Nd_a + d_1}{N + 1}$ is the average density of the mixture.

Notice that the momentum of the gas is sufficient for the non-
friction condition. As soon as friction effects are considered,
things change. Also, to be more realistic, one must consider
the vena contracta of the nozzle and the available momentum for
entrainment. Let us redefine v_1 as the actual velocity of the
vena contracta.

Let $e_1 \, H_1$ be the actual pressure head in the gas stream
which is available for frictionless flow, and let $e_2 \, H_2$ be the
actual pres ure head available in the mixture for frictionless
flow. The volume of gas/sec is given by

$$q = a_1 \, v_1$$

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and the mixture by

$$q(N + 1) = a_2 \, v_2$$

where a_1 and a_2 are the areas of the vena contracta and throat respectively. The momentum in the gas stream is

$$M_1 = m_1 \, v_1 = a_1 \, v_1^2$$

and

$$M_2 = m_2 \, v_2 = \frac{(Nd_a + d_1)}{N + 1} \, a_2 \, v_2^2$$

Now $\dfrac{v_1^2}{2g}$ is the velocity head of the gas

and $\dfrac{v_2^2}{2g}$ is the velocity head of the mixture

The velocity head available from the gas is given by

$$\frac{M_1}{2g} = d_1 \, a_1 \, e_1 \, \frac{v_1^2}{2g}$$

and from the mixture

$$\frac{M_2}{2g} = \frac{(Nd_a + d_1)}{N + 1} \, a_2 \, e_2 \, \frac{v_2^2}{2g}$$

These two equations represent the available heads as if there were frictionless conditions. Of course, e_1 and e_2 actually account for friction losses.

Then,

$$d_1 \, a_1 \, e_1 \, v_1^2 = \frac{(Nd_a + d_1)}{N + 1} \, a_2 \, e_2 \, v_2^2$$

but $\dfrac{q_1}{a_1} = v_1$ and $\dfrac{q_2}{a_2} = v_2$

so,

$$d_1 \, a_1 \, e_1 \, \frac{q_1^{\,2}}{a_1^{\,2}} = \left(\frac{Nd_a + d_1}{N \quad 1}\right) a_2 \, e_2 \, q_1^{\,2} \, \frac{(N + 1)^2}{a_2^{\,2}}$$

Reducing this expression gives:

$$e_1 \frac{d_1}{a_1} = e_2 \frac{\left(Nd_a + d_1\right)}{a_2}(N + 1)$$

and

$$\boxed{\frac{a_1}{a_2} = \frac{e_1}{e_2}\left[\frac{d_1}{Nd_a + d_1}\right]\frac{1}{N + 1}}$$

Eq. (8)

This theoretical expression can be used to predict entrainment ratio, once the efficiencies e_1 and e_2 are determined, and the densities are known. Notice that if compressed air is used for the nozzle gas and air is entrained, then the densities cancel out and

$$\frac{a_1}{a_2} = \frac{e_1}{e_2}\left(\frac{1}{N + 1}\right)^2$$

Eq. (9)

This is a very simple expression to test once e_1 and e_2 are known. The efficiency e_1 is related to the nozzle flow coeffic- ient and the nozzle friction by

$$e = C^2 = \frac{1}{1 + f} \qquad \text{as previously stated}$$

The flow coefficients for various nozzles are well documented in the fluid flow literature, so it will be assumed that e_1 is known for a particular nozzle in use.

Equation (8) assumes that both gases are at the same temper- ature but have different densities. Equation (9) assumes equal densities. Either one of these equations can be used to compare test results with theory.

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3.0 DESIGNS

3.1 Air Ejector Design

The design chosen for the air ejector was based on its possible use as a truck exhaust pipe. The ejector was hogged out of solid aluminum for these experiments. The entrance to the throat was machined to form a circular bell mouth of 14" radius. The mixing tube had a diameter of 3". The expander section expanded to 4" diameter at a total included angle of 4.76°. A four inch diameter pipe was attached to the expander. The pipe length was 30" to assure good turbulent mixing.

3.2 Nozzle Design

A scaled up version of nozzle number 4 (Figure 4) was used. Two nozzles were tested, one with a 0.75" diameter and the other with a 0.5" diameter orifice. The angle of approach of both nozzles was 10°.

After carefully reading the technical literature, it becomes clear that a channel type orifice is not efficient for air entrainment. The sharp edged orifice is better. There are many types of sharp edged orifices and all of them are about equal for entrainment purposes. However, they are not equal as far as flow efficiency is concerned. Consider the following orifice designs.
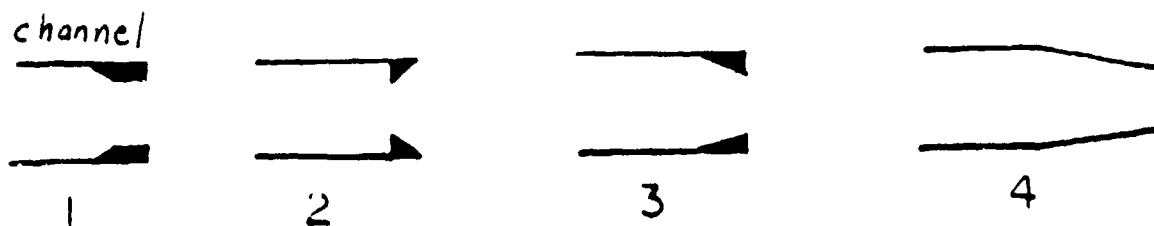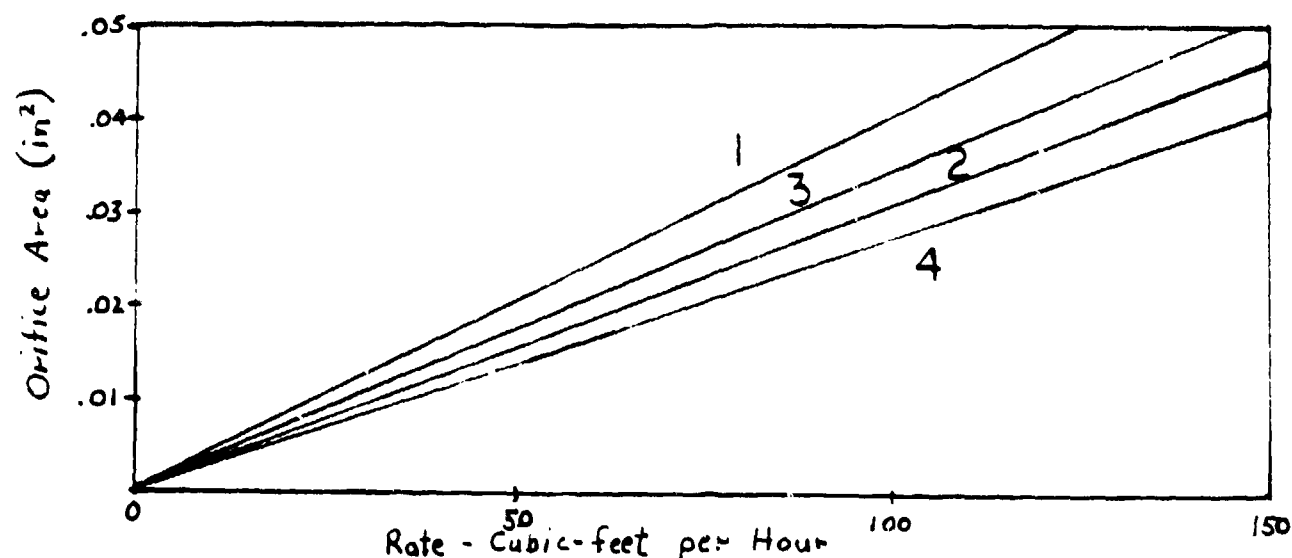
channel

1   2   3   4

Figure 4

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The literature shows that numbers 2, 3 and 4 are equal from an entrainment viewpoint and superior to the channel orifice. The literature shows also that orifice number 4 has the best efficiency. Consider some experimental data from the National Bureau of Standards.

Figure 5

We will use a scaled up version of nozzle number 4. The first experiment will use a nozzle of 1 inch diameter reduced to an orifice of 0.75 inch diameter.

4.0 EXPERIMENTAL PROCEDURES

4.1 Apparatus

The experiments were carried out on a heavy duty optical bench. Figure 6 shows that the ejector was mounted on a moveable platform and the exhaust pipe was mounted on another moveable platform. The drawing shows that a heater was directed into a plenum where it heated the air in the plenum. This air was drawn out from the plenum by a high pressure blower and the

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FIGURE #6

DIFFERENTIAL MANOMETER

AIR VELOCITY METER

ORIFICE FLANGE PLATE ADAPTOR
CONTROL VALVE
MOUNT JIG CARRIER
LONGITUDINAL ADJUSTMENT
2" ID
REDUCER
NOZZLE
1" ID
VENTURI
MOUNT. JIG CARRIER
LATERAL ADJUSTMENT
OPTICAL BENCH
EXTENSION
FLEXIBLE STEEL TUBING
I.H.P. BLOWER
PLENUM
PITOT TUBE
STAGE & STAND
PORTABLE HEATER
30,000 BTU
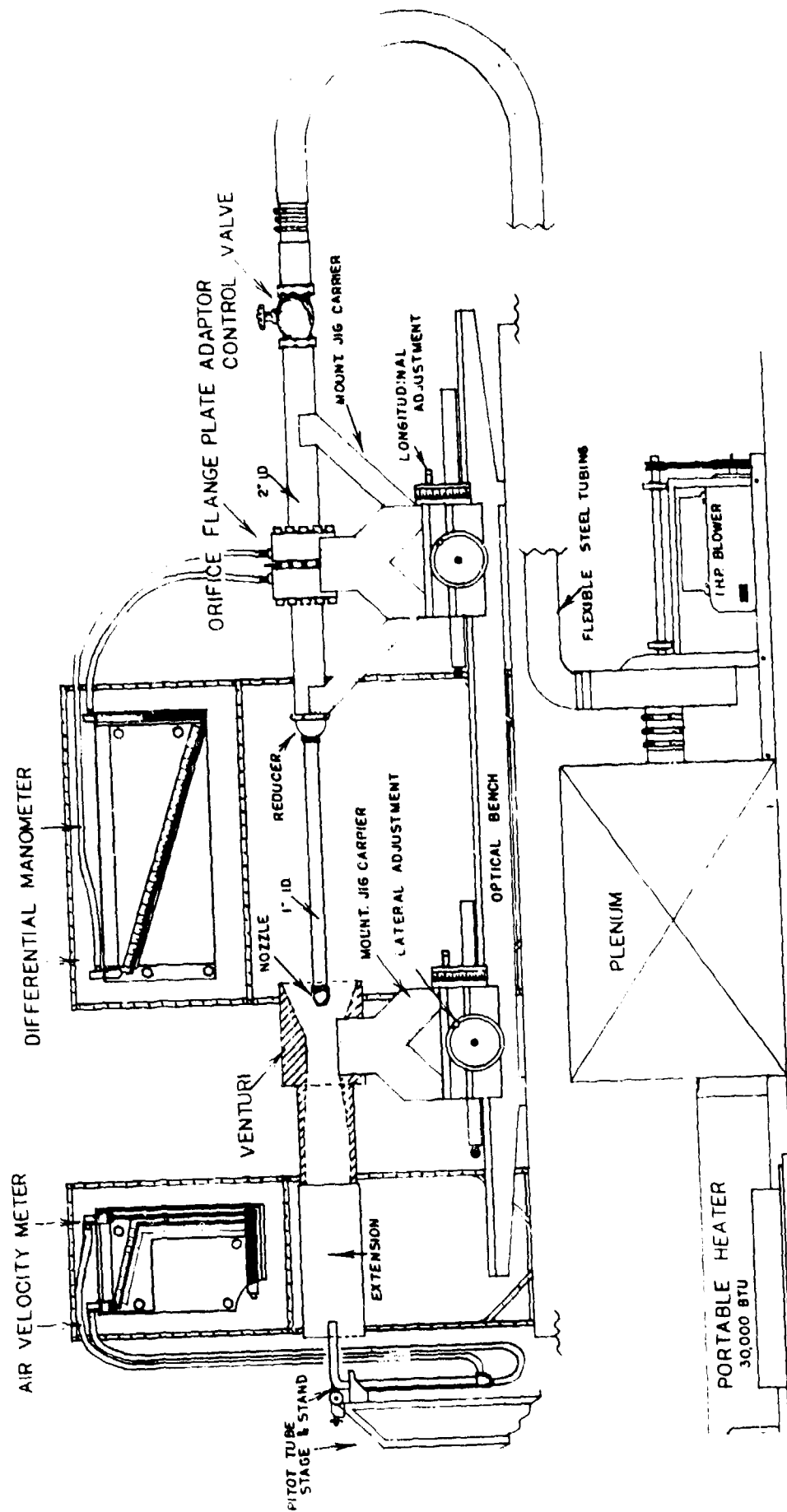
AREA THERM. CORP.

| N/A | | |
| 12 JULY 82 | | |

THERMAL SUPPRESSOR – EXHAUST

ENTRAINMENT TEST BED | 3-0040

high pressure hot air was directed through a control valve and then through an orifice plate. The orifice plate was used to measure the air flow in the pipe. The hot air was forced through a nozzle which was screwed onto a one inch pipe. The hot air leaving the nozzle was directed into the throat of the air ejector, where it entrained cool air from the outside. This mixture was directed into a long pipe (this would be an exhaust pipe) where its flow was measured by a pitot tube.

4.2 Measurement of Nozzle Flow

The nozzle flow was measured by using a calibrated orifice plate and a differential manometer. The manometer was read to 0.02" H_2O. Published tables give the flow coefficient, C, of the orifice plate as C = 0.63. The flow Q_1 through the orifice (and through the nozzle) is given by

$$Q_1 = CA \sqrt{\frac{2\Delta p}{\rho}} = 6C\ D^2 \sqrt{\frac{h}{0.075d}}$$

where C = flow coefficient of the orifice

A = area of the orifice

Δp = differential pressure
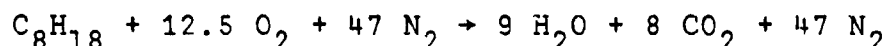
ρ = density of the air

D = diameter of orifice

h = differential water height

d = correction for density

The only term not known directly was the air density. Air density can be calculated if the air temperature, relative

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humidity and barometric pressure are known. However, when the
flow is from an engine or kerosene heater, the vapor partial
pressure is not the same as the ambient partial pressure. Con-
sider the burning of octane which is chemically close to the
burning of diesel fuel.

$$C_8H_{18} + a\ O_2 + 3.76\ a\ N_2 \rightarrow b\ H_2O + C\ CO_2 + 3.76\ a\ N_2$$

$$C_8H_{18} + 12.5\ O_2 + 47\ N_2 \rightarrow 9\ H_2O + 8\ CO_2 + 47\ N_2$$

The total mass

$$162 + 352 + 1316 = 1830\ lbs.$$

The total moles

$$9 + 8 + 47 = 64$$

The partial pressure of 9 H_2O in 64 moles of gas is

$$P_v = \left(\frac{9}{64}\right)\ 14 = 1.97\ psia$$

This value of vapor pressure was used for density calculations
whenever the heater or engine was used as the input to the
nozzle. For those experiments using air, density was calculated
or read from an air flow slide rule.

4.3 Measurement of Entrained Air

The entrained air was measured by an air velocity meter.
This meter utilized a pitot tube that measured total pressure
and static pressure. The pressure difference was measured to
0.01" H_2O. The velocity reading was corrected for air temper-
ature and air density by using a Dwyer air flow slide rule.
Since the entrainment ratio was always greater than 4, the vapor
pressure for ambient air was used to determine air density. The

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entrained flow Q_2 was calculated by using

$$Q_2 = A_2 <v>$$

where $<v>$ is the average velocity in the entrainment pipe

The average velocity was obtained by taking several readings across the entrainment pipe and then averaging these readings.
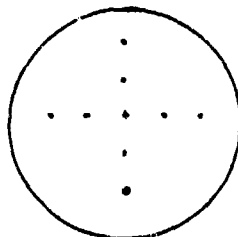
At times there was very little difference in the readings. Whenever there were large differences, more positions were measured.

4.4 Changing of Air Flow

The air flow for both heated and non-heated air was changed by opening or closing a valve. The exhaust flow from most small engines is from 10 ft^3/min to 20 ft^3/min. The manifold was designed to accomodate these values.

5.0 DATA

5.1 Air Flow Data

Three basic experiments were performed on both the 0.5" diameter nozzle and the 0.75" diameter nozzle;

(1). measurements for unheated air

(2). measurements for heated air (kerosene heater)

(3). measurements for a 3kW E-G set

5.1.1 Unheated Air Experiments

Date: 5 Aug 82
Place: Area Therm Corp., Springfield, Va.

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Air Temperature: 82°F

Relative Humidity: 68%

Barometric Pressure: 29" Hg.

Calculated Air Density = 0.07 lbs/ft^3

1/2" Nozzle

$$Q_1 = 3.78\sqrt{\frac{h}{0.07}} = 14.3\sqrt{h}$$

$$Q_2 = 0.098\ \langle v_2 \rangle$$

Q_1 \ h	2.14	1.84	1.6	1.18	0.7	0.24
Q_1	21	19.4	18.1	15.5	12	7
$\langle v_2 \rangle$	2280	2150	1970	1620	1150	590
Q_2	223	211	193	158	113	58
$\dfrac{Q_2}{Q_1}$	10.6	10.8	10.7	10.2	9.4	8.3

0.75" Nozzle

Q_1 \ h	5.47	4.36	3.41	2.38	1.44	0.38
	33.4	29.8	25.3	22	17.1	8.8
v_2	2324	2080	1780	1470	1060	415
Q_2	227	204	174	144	104	40.6
$\dfrac{Q_2}{Q_1}$	6.8	6.8	6.9	6.5	6.1	4.6

5.1.2 Heated Air Experiments

(Same conditions as unheated air experiments)

1/2" Nozzle

$$Q_1 = 3.78\sqrt{\frac{h}{.066}} = 14.7\sqrt{h}$$

$$Q_2 = 0.098\ \langle v_2 \rangle$$

h	1.73	1.16	0.6	.28
Q_1	19.4	15.8	11.4	7.8
$\langle v \rangle$	2100	1600	960	430
Q_2	206	157	94	42
$\dfrac{Q_1}{Q_2}$	10.6	9.9	8.3	5.4

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0.75" Nozzle

h

	4.98	4.04	2.92	.95	.37
Q_1	32.8	29.6	25.1	14.3	8.9
$<v>$	2340	2050	1680	790	430
Q_2	229	201	165	77.4	42
$\dfrac{Q_2}{Q_1}$	7	6.8	6.6	5.4	4.7

5.1.3   3kW Exhaust Experiments

> Date: 4 Aug 82
> Place: Area Therm Corp., Springfield, Va.
> Air Temperature: $90^{\circ}F$
> Relative Humidity: 46%
> Barometric Pressure: 29" Hg.

5.1.3.1   Experimental Set-up

For this experiment, the air flow manifold was used to mea-
sure the flow from one exhaust pipe of the 3kW engine-generator
set. The very hot temperature of the exhaust pipe could not be
transferred down the manifold with pipe insulation. But even
with pipe insulation only $125^{\circ}C$ gas was incident at the measuring
orifice. However, these experiments were used to measure flows
from the 3kW and also back pressures.

The measured back pressure of the 3kW with its muffler was
0.5" H_2O. When the 0.75" nozzle was tested, the back pressure
went to 1.5" H_2O. When the 0.5" nozzle was used, the back pressure
went to 2.5" H_2O. The muffler was not on the set when the nozzles
were tested.

5.1.3.2 Data

The temperature of the exhaust gas at the measuring orifice was 257^OF. The calculated density for this temperature and for the kerosene vapor partial pressure was

$$\rho = 0.06 \text{ lbs/ft}^3$$

for 1/2" nozzle

$$Q_1 = 3.78\sqrt{\frac{h}{\rho}} = 3.78\sqrt{\frac{0.89}{0.06}} = 14.6 \text{ ft}^3/\text{min}$$

$$Q_2 = 0.098 <v> = 0.098 \times 1160 = 114$$

$$N = \frac{Q_2}{Q_1} = \frac{114}{14.6} = 7.8$$

for 0.75" nozzle

$$Q_1 = 3.78\sqrt{\frac{1.3}{0.06}} = 17.6$$

$$Q_2 = 0.098 \times <1240> = 121.5$$

$$N = \frac{Q_2}{Q_1} = 6.9$$

5.2 Temperature Data

The final goal of an exhaust suppressor is to lower the temperature of the suppressor itself. The suppressor is actually "hiding" the engine's exhaust pipe. Three temperature experiments were done for both nozzles:

(1). use of hot air from the heater

(2). use of exhaust gas through the manifold

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(3). use of exhaust gas directly fed into the ejector
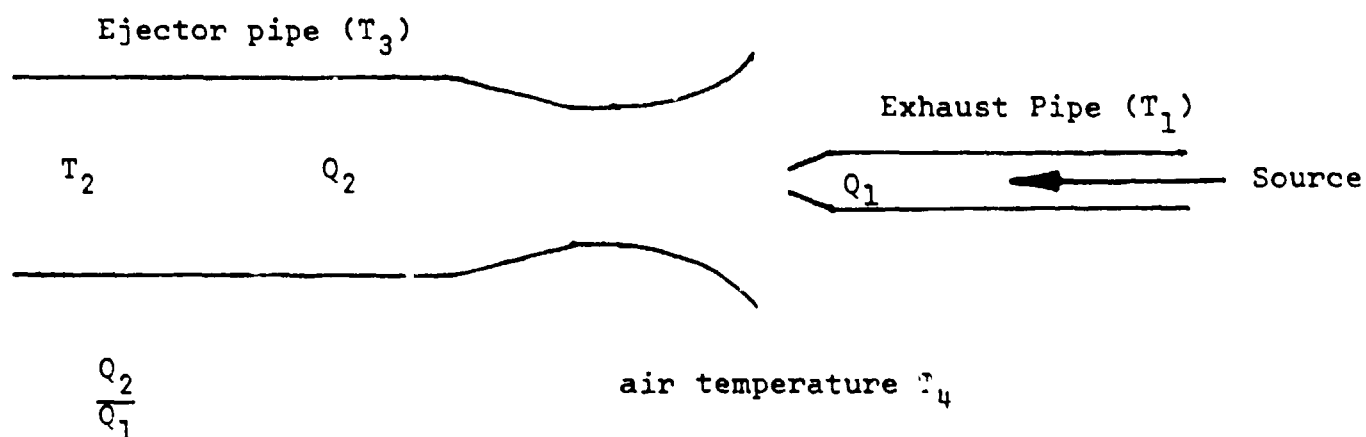
Drawings help to explain the tests and the data.

Ejector pipe (T_3)

T_2 Q_2

Exhaust Pipe (T_1)

Q_1 ⟵   Source

$\dfrac{Q_2}{Q_1}$ air temperature T_4

Figure 7

There was no significant difference between the 1/2 inch and the
0.75 inch nozzle

| source	nozzle	$\dfrac{Q_2}{Q_1}$	T_1	T_2	T_3	T_4	$T_1 - T_3$
heater	.5	10.6	75	40	39	31	$36C^{o}$
	.75	7	80	42	41	31	$39C^{o}$
exhaust & manifold	.5	7.8	104	41	38	32	$63C^{o}$
	.75	6.9	91	42	39	32	$53C^{o}$
direct exhaust	.5	10.7	436	55	50	27	$386C^{o}$
	.75	5.9	381	52	49	27	$332C^{o}$
	1.0*		340	56	53	27	$287C^{o}$

*1.0 represents the no nozzle case

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Figure 8 shows the flow data on graph paper. The abscissa is Q_1, the flow through the nozzle and the ordinate is entrainment ratio.

5.3 Comparison of Load and No-load Conditions

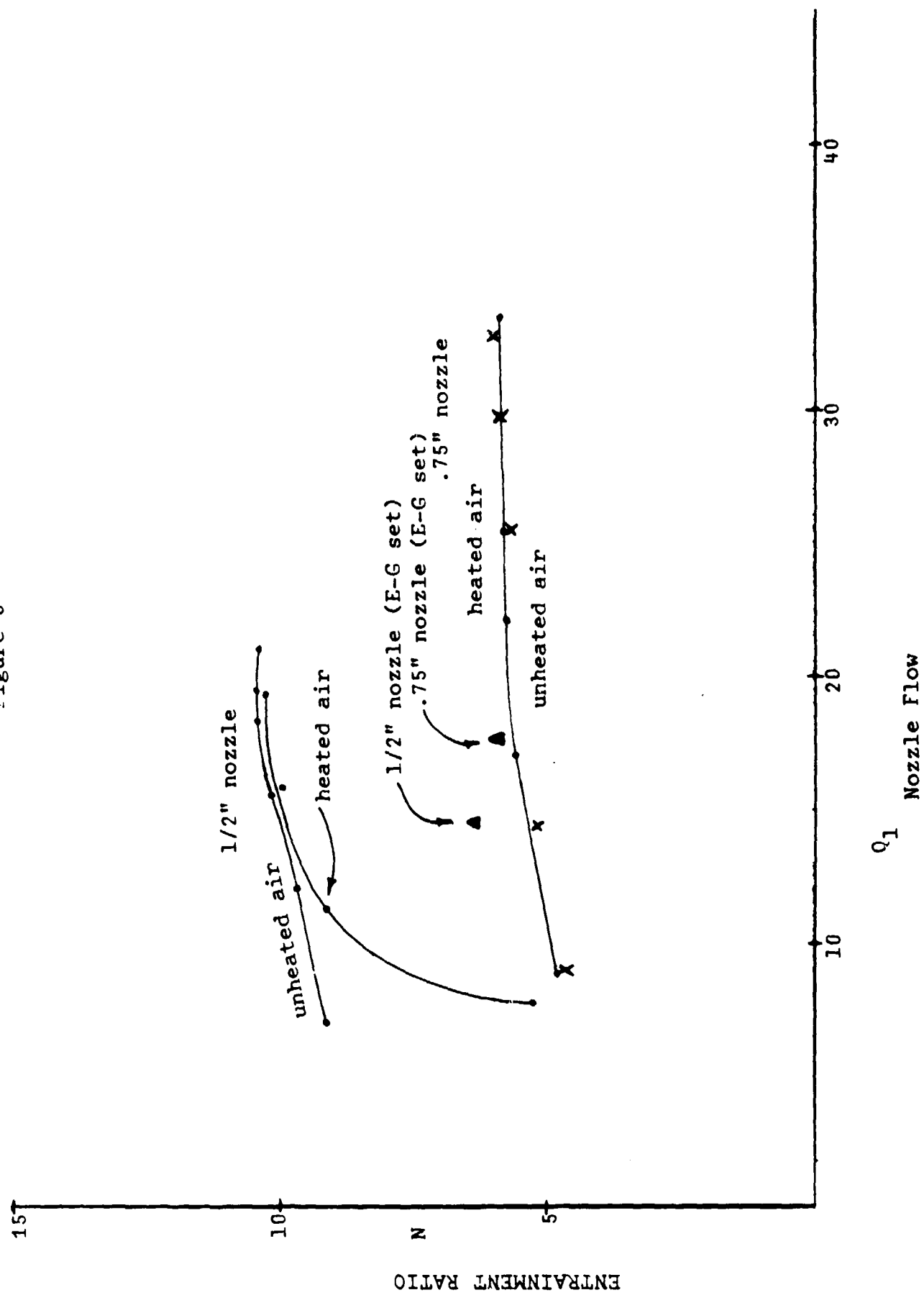The E-G exhaust was fed directly into the ejector for full electrical load and no electrical load.

Figure 9
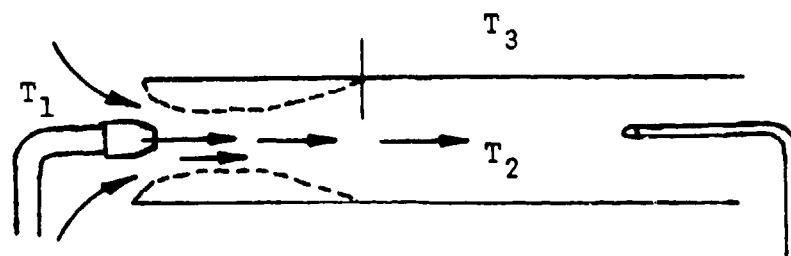
air = 28°C

Nozzle		T_1 ($^{\circ}$C)	T_2($^{\circ}$C)	T_3($^{\circ}$C)	$T_1 - T_3$
none	load	433	76	70	363
	no load	357	58	51	306
3/4"	load	409	63	58	351
	no load	370	50	45	325
1/2"	load	415	61	59	356
	no load	393	45	43	350

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Figure 8

6.0 RESULTS

6.1 Airflow

(1). For air-to-air conditions, i.e., air as the driving force and air entrained, the best entrainment ratios were

    .for the 1/2 inch nozzle, $N = 10.6$
    .for the 0.75 inch nozzle, $N = 6.9$

(2). For heated air-to-air conditions, the best entrainment ratios were

    .for the 1/2 inch nozzle, $N = 10.6$
    .for the 0.75 inch nozzle, $N = 6.9$

(3). As the flow Q_1 decreased, the entrainment ratio decreased for both heated and unheated air.

(4). For direct E-G exhaust into the ejector, the entrainment ratios were

    .for the 1/2 inch nozzle, $N = 10.5$
    .for the 0.75 inch nozzle, $N = 5.9$

(5). The use of a nozzle definitely increased the entrainment ratio. Comparison of temperature reduction of nozzle vs non-nozzle clearly shows the effect of nozzles on temperature reduction.

6.2 Temperature Reduction

(6). Very large temperature reductions were measured. $\Delta T = T_1 - T_3$, i.e., the difference between the engine exhaust pipe and the ejector pipe, was greater than $350C^{\circ}$ for most cases.

    .The ejector pipe without a nozzle had the highest temperatures.

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.The 1/2" and 0.75" nozzles were very close
in performance for temperature reduction.

7.0 CONCLUSIONS

(1). From a temperature viewpoint, the experiments were
very successful. Dropping the temperature from
around 400°C to 50°C must be considered as an out-
standing success from the viewpoint of thermal
suppression.

(2). Entrainment ratios from 6 to 10 were better than
theoretical predictions.

(3). The reduction in entrainment ratio, as the primary
flow was reduced, was significant for the 1/2"
nozzle but not for the 0.75" nozzle. This may be
due to a loss in nozzle efficiency for low exit
velocities.

(4). The importance of a nozzle is clearly demonstrated
by temperature reduction data.

(5). For the 3kW E-G set, the 0.75" nozzle was about the
same as the 0.5" nozzle from the thermal suppression
viewpoint. The 0.5" nozzle created more back pressure
than the 0.75" nozzle, and raised the exhaust temper-
ature. Even though entrainment was better for the
1/2" nozzle, the resulting surface temperature of
the ejector pipe was not significantly better.

8.0 FINAL DISCUSSION

The basic concept of using a nozzle on an engine's exhaust
pipe and directing the exhaust into an air ejector is clearly very

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useful for thermal suppression. For these experiments, a one
inch exhaust pipe was directed into a three inch ejector mixer.
Theory predicts and experiments prove that the entrainment
ratio increases as

$$\text{ratio} = \frac{\text{area of mixer}}{\text{area of nozzle}} \qquad \text{increases.}$$

For example, if the mixer area were 5" instead of 3", the ratios
would become

$$\text{ratio} = \left(\frac{5}{.75}\right)^2 = 44 \qquad \text{instead of} \qquad \left(\frac{3}{.75}\right)^2 = 16$$

This means that a significant increase of entrainment ratio could
cool the ejector pipe down close to room temperature. This would
be a complete solution to the exhaust pipe problem. The only
disadvantage to increasing the diameter of the ejector pipe is
that its bigger.

When one considers that the temperature reduction was
accomplished by using a single piece of pipe, the results are
astounding. Such pipes could be manufactured at very, very low
costs. This cost-performance value is hard to beat.

9.0 RECOMMENDATIONS

(1). It is clear now that the use of air ejectors to cool
exhaust pipes is a very successful technique. Out-
standing results were obtained for one design only.
Obviously, there is a possibility that the results
might be even better for other designs. The field
is wide open now. Therefore, it is suggested that
the following investigations should be undertaken:

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.flow characteristics of various nozzles
under real exhaust gas conditions. Note
that the characteristic of interest is
__entrainment__ efficiency. This type of
information is not available in the
literature. A new nozzle concept may be
required for optimum performance.

.efficiency of entrance shapes for entrain-
ment. The shape of the "bell mouth" is
very important, and very little information
is available in the technical literature.

.effects on efficiency due to curved caps
and due to muffler covers
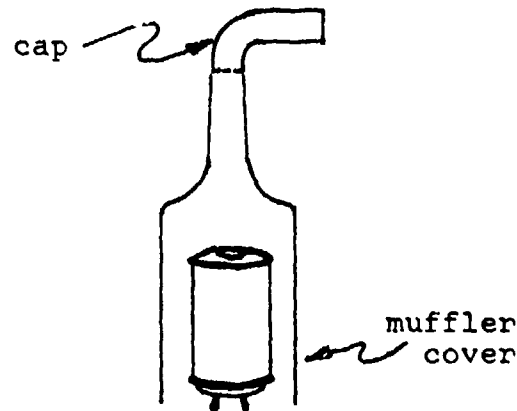
(2). A test program on thermal characteristics of engines
due to back pressure should be undertaken. From the
test data it was shown that entrainment increased as
nozzle diameter decreased but back pressure caused
increased engine temperature. There may be a very
interesting trade-off between nozzle size and increased
engine temperature due to the increased back pressure.

(3). There is a need for large scale experiments, especially
with turbines. Large scale systems should perform as
well as smaller systems, but experimentation is needed.

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE	READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER DM-23.1	2. GOVT ACCESSION NO. AD-A119525	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) NAVFAC Design Manual DM-23.1 Airfield Lighting		5. TYPE OF REPORT & PERIOD COVERED Design Criteria Final
		6. PERFORMING ORG. REPORT NUMBER DM-23.1
AUTHOR(s) Naval Facilities Engineering Command 200 Stovall Street Alexandria, VA 22332 (Code 0442)		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS Naval Facilities Engineering Command 200 Stovall Street Alexandria, VA 22332		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS Engineering and Design
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Facilities Engineering Command (Code 0432) 200 Stovall Street Alexandria, VA 22332		12. REPORT DATE July 1981
		13. NUMBER OF PAGES 42
14. MONITORING AGENCY NAME & ADDRESS(If different from Controlling Office)		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)
Unclassified/Unlimited

This document has been approved

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)
Unclassified/Unlimited

18. SUPPLEMENTARY NOTES

SEP 23 1962

A

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)
Airfield lighting (general); airfield lighting systems; carrier deck
lighting (simulated); common lighting criteria; helipad lighting;
parking/service area lighting; runway lighting requirements.

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)
Design criteria are presented for Category Code 136 and are intended for use
by experienced architects and engineers. The contents include approach
lighting, runway lighting, taxiway lighting; and helipad lightings.

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1 JAN 73